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DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

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EVALUATION OF VARIOUS ROTARY MECHANISMS FOR POTENTIAL AS HIGH-PRESSURE SHIPBOARD COMPRESSORS

By
William Thelen

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PROPULSION AND AUXILIARY SYSTEMS DEPARTMENT

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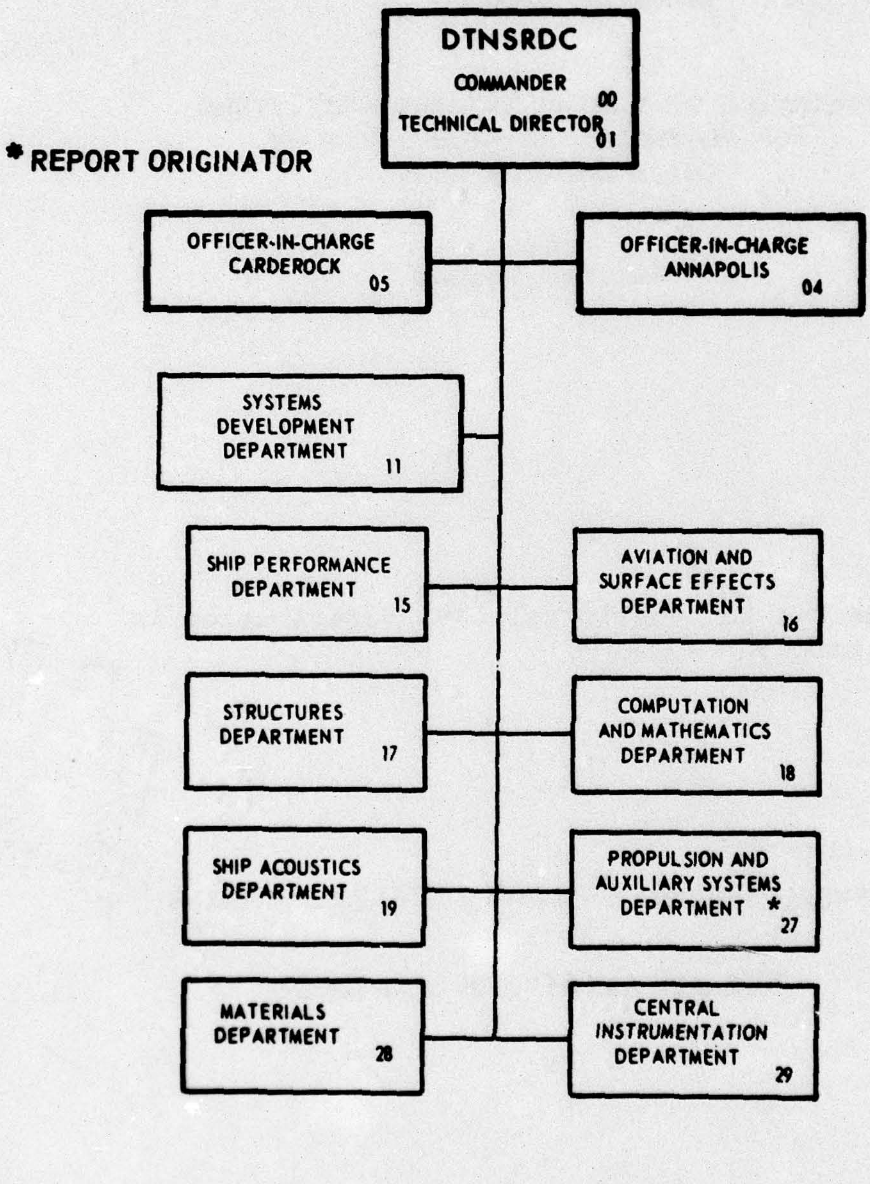

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ADMINISTRATIVE INFORMATION

The work described herein was accomplished under Work Unit 2745-541 and constitutes milestone 4 of the 1 August 1976 Laboratory Program Summary. Dr. F. Ventriglio, NAVSEA (SEA 0331F), was the program manager and Mr. V. Wiersma, NAVSEC (SEC 6153B), was the program technical agent.

LIST OF ABBREVIATIONS

atm	- atmosphere
cfh	- cubic feet per hour
cm	- centimetre
cm^3/min	- cubic centimetre per minute
°C	- degree Celsius
°F	- degree Fahrenheit
°K	- degree Kelvin
E_m	- mechanical efficiency
E_v	- volumetric efficiency
f/m	- feet per minute
hp	- horsepower
kg	- kilogram
kPa	- kilopascal
kw	- kilowatt
kg/min	- kilogram per minute
m^3/hr	- cubic metre per hour
mm	- millimetre
m/s	- metre per second
ROH	- rotary-piston, oil-free, high-pressure (compressor)
psi	- pounds per square inch (absolute)
lb/hr	- pound per hour
scfm	- standard cubic feet per minute
WFHS	- water-flooded helical-screw (compressor)
A	- area (cm^2)
C	- P/P_1 (-)
D	- gap between rotor and housing at discharge (cm)
E	- efficiency (-)
f	- coefficient of friction (-)
g	- acceleration due to gravity (9.807 m/s^2)
G	- gas constant (1.986 calorie/kg - °K)
h	- clearance (mm)
K	- polytropic coefficient (-)
L	- length (cm)
M	- Mach number (-)
m	- mass flow rate (kg/min)
N	- number of pockets (-)
P_2	- downstream pressure (atm)
P_1	- upstream pressure (atm)
r/m	- revolutions per minute

s	-	second
P	-	pressure (atm)
R	-	radius (cm)
R_i	-	rotor radius
R_3	-	radius of clearance seal between housing and rotor (cm)
R_o	-	housing radius
R_4	-	radial length of vane (cm)
S	-	mechanical loss (kw)
T	-	temperature (°K)
T_1	-	upstream temperature (° Kelvin)
u	-	absolute viscosity (kg-s/m^2)
V	-	velocity (m/s)
X	-	factor obtained from appendix A (-)
X'	-	effect of moving wall on flow rate (-)
X''	-	ratio of leakage of water and leakage of air (-)
θ	-	angular position (radians)
<	-	less than

SUBSCRIPTS

t	-	total
o	-	outside or housing
i	-	inside or rotor
v	-	volumetric
m	-	mechanical
L	-	loss
f	-	final
1	-	original (inlet)
a, b, c, d	-	counting indices for identification only

TABLE OF CONTENTS

ABSTRACT

The suitability of a number of rotary mechanisms for potential use as 200-atmosphere (3000 pounds per square inch), oil-free shipboard air compressors was investigated. The structural, operational, and manufacturing limitations were analyzed and the expected efficiencies calculated. The Zimmern single-screw compressor appears to be a feasible and technically attractive candidate, especially if water-lubricated bearings are employed. All other mechanisms were judged to be unsuitable for the required service or to require the development of technology well beyond that which is now available.

INTRODUCTION

BACKGROUND

Air which is required aboard ship at pressures above 10 atm* (150 psi) is currently provided by multistage reciprocating piston compressors rated at 200-350 atm (3000-5000 psi). Even the most modern of these compressors weigh up to 3300 kg (7500 pounds), generate considerable structureborne vibration, and, due to the large number of rubbing parts, have limited reliability and service lives.

During the last 10 years, Lysholm twin-screw compressors have demonstrated dramatic advantages in size, noise, maintenance, and logistics over reciprocating piston compressors for 10 atm air service. These advantages resulted from the reduction in the amount of ancillary apparatus and moving parts, the elimination of valving, and the lowering of discharge temperatures. Recently, WFHS compressors have demonstrated the same mechanical advantages while providing oil-free air. They have been selected for use on FFG-7 class ships. On the basis of that achievement, it would be highly desirable to obtain the same advantages for high-pressure service by using an ROH to meet shipboard air requirements between 10 and 300 atmosphere. Unfortunately, since there is little commercial requirement for a small ROH, there is little incentive for commercial development of a shipboard ROH.

There are many rotary mechanisms discussed in the literature which have been proposed for use as engines or low-pressure

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*Definitions of abbreviations appear on page i.

compressors, but due to lack of commercial interest their suitability as ROH compressors has not been addressed.

SCOPE

This report covers an examination of the potential of rotary positive displacement mechanisms and dynamic concepts as a ROH compressor for shipboard service.

APPROACH

The literature was searched, and commercial compressor designers were contacted to compile the list of candidates and to establish the procedures for their evaluation. Only facets which appeared to limit the compressor utility were examined. Generally, the highly loaded components were examined first to see if they could sustain, or be modified to sustain, the required operating pressures. The volumetric and mechanical efficiencies of the structurally sufficient candidates were then calculated. Since the structural and efficiency calculations are related, the procedure, in some cases, evolved into an iteration process. Finally, the most promising designs were examined more closely to see what technology would be required to design and fabricate the required compressor. The conditions shown in table 1 were postulated to represent those of the last stage of a shipboard high-pressure compressor and were used throughout.

TABLE 1
DESIGN CONDITIONS

Discharge Capacity, kg/min	3.62 (480 lb/hr)
Discharge Pressure, atm	204 (3000 psi)
Inlet Pressure, atm	68 (1000 psi)
Inlet Air Temperature, °K	321 (120° F)
Input Shaft Speed, r/m	3600

Note: The discharge capacity shown is equivalent to 30 cfh (compressed cubic feet per hour) at 3000 psi and was selected because it represents the largest high-pressure compressor capacity in the Fleet today.

Lower pressure stages were not examined at this time because it was reasoned they would present an easier problem and would probably utilize a spin-off of the high-pressure design. Also, no limitations were placed on how such ROH technology should be applied to future shipboard needs.

The large number of designs and the complexity of their geometry made an exact analysis impossible. The assumptions listed below are employed wherever more valid information was unavailable.

- Leakage does not affect pressure distribution.

- During the compression process, volumes are assumed to decrease at a uniform rate.

- Isothermal compression results from the intimate contact of product air with cooling/sealing water.

- Cooling water does not affect volumes.

- Leakage was calculated as described in appendix A.

- Air is assumed to be a perfect gas.

- Suction and discharge pressure are constant.

- All moving seals have a clearance of 0.001 times the appropriate rotor dimension.

- Internal nonworking volumes are vented to suction.

Although these assumptions will not be completely valid, it is believed that deviations that can be expected to occur will not significantly affect the results.

Because of the simplicity of its design and the resulting calculations, the sliding-vane compressor was used to estimate the relative importance of various losses under different operating conditions. These calculations are included in appendix B to demonstrate the methods used throughout. In some cases the results of these calculations were used to estimate the mechanical losses of machines with more complex geometry. Since the intent of this work was to evaluate the potential of various mechanisms, considerable freedom was taken to determine the effect of modifying existing designs to accommodate the higher pressures or to incorporate improved bearing and sealing designs. All of these modifications, however, involve available technology.

RESULTS

SUMMARY

The results of this investigation are summarized in table 2. The analysis of the various compressor/engine mechanisms shows the Zimmern single-screw compressor design to be the most highly promising candidate for small capacity, high-pressure service. It appears that its conversion to the desired service can be accomplished in a relatively straightforward manner through the application of advanced, but available, technology. With the

application of considerably more development effort, the designs in group II could probably be developed to meet the required operating characteristics. These designs, however, would require extensive development and modification to support the high operating pressures. Based on this investigation, the resulting machines would likely be less efficient, more complicated, and generally less attractive than the Zimmern which could be developed at a fraction of their cost.

TABLE 2
SUMMARY OF RESULTS OF INVESTIGATION

Design	Structural Limitation (Maximum Discharge Pressure)	Efficiency, %[1] Volumetric E_v	Mechanical E_m	Required[1] Rotor Diameter cm	Advantages[2]	Disadvantages	Required Technology	Comment
				Group I - Technically Attractive				
Zimmern (single screw)	Gaterotor deflection (above 300 atm).	72	90	5	No timing gears. No large pressure forces to support.	Some sliding contact (not pressure actuated).	Design. Manufacturing	
				Group II - Potentially Suitable But Requiring Considerable Development				
Wankel (driven housing)	Bearing load (about 200 atm).	65	80	7	Developed commercial technology.	Difficult porting. Requires timing gears.	Air Seal. Oil seal. Overall design.	
Clarke	Bearing load (about 200 atm).	46	55	4		Difficult fabrication. Large loads on bearings.	Design. Fabrication.	
Clarke (Tandem mount)	Rotor deflection.[3]	46	90	4		Difficult fabrication.	Design. Fabrication.	Double capacity.
Scroll	Rotor deflection (above 300 atm).[3]	80	75	5		Large seal area. Requires rubbing air seals.	Design. Seal and bearing materials.	Calculations assume contacting seals. Is attractive for oil lubricated service.
Disk	Rotor stress.[3]	(5)	-(4)	-	Potentially low fabrication cost. No unbalanced pressure.	Requires 20,000 to 30,000 r/m operation.	Fabrication. Design	
Centrifugal	Rotor stress.[3]	(5)	-	-	Sophisticated design capability available.	Poor efficiency anticipated. Requires above 100,000 r/m. Many stages of compression.	Fabrication. High speed drive.	Results taken from general literature.
				Group III - Not Suited for This Service				
Lysholm (twin screw)	Bearing loads (50 atm).	60	50	15	Oil-free technology available.	Precision required during overhaul.		8 lobe/12 lobe rotor design.
Wankel (driven rotor)	Bearing loads (75 atm).	65	80	7		Nonrotary motion.		
Sliding Vane	Vane stress and vane friction (20 atm).	52	56	8				
Liquid Ring	(15 atm).[6]	-	-	-				
Tschudi	3	20	-	9		High inertia loads.		Torus diameter of 20 cm.
Unsing	Bearing load. (-)	Less than 20	-	15				
Reciprocating piston	None	60	More than 85	3.5				26 cm stroke. Included for comparison only, calculated with clearance seals.

[1]Assumes conditions in table 1. Structural limitations are neglected for group III.
[2]Does not include advantages common to ROH compressors.
[3]No well-defined limit. Increased pressure capability purchased at expense of efficiency.
[4] - = not calculated.
[5]Not a positive displacement machine.
[6]Structural parts are lightly loaded. Maximum pressure is related to fluid losses.

It should be noted, however, that for oil-lubricated service, the scroll compressor (shown in group II, table 2) equipped with pressure-actuated strip seals could probably be developed into an efficient and reliable compressor with a relatively small-scale effort. The analysis of the individual concepts are discussed below.

ZIMMERN COMPRESSOR

The single-screw compressor[1] shown in figure 1 was developed by Zimmern about 15 years ago. It is used commercially to compress air, in one stage, to pressures as high as 18 atmosphere. The compressor consists of a single hour-glass-shaped screw which cooperates with two symmetrically located free-spinning gaterotors to form compression pockets. The pockets are closed and the gas compressed prior to radial discharge into the housing. Unlike the sliding-vane or Wankel, the pocket is decreased in three dimensions and there is virtually no undelivered volume. (The Wankel reduces only the radial dimension of its pocket.) This, coupled with the absence of the "blow hole" present in twin-screw compressors and the ability of the gaterotor, to follow the rotor, provides a unit with excellent volumetric efficiency. The efficiency of 93% reported with commercial machines[1] is about 7% better than that of any other commercial rotary compressor.

The symmetry of the design allows the rotor to be free of both axial and radial forces. Also, since the gaterotor must only support the pocket pressure over the area that extends into the pocket, its bearing loads and stress levels are relatively low. The deflection of the gaterotor teeth appears to be the limiting structural factor with this design. Thicker gaterotors would be required for higher pressures than used in commercial machinery, but this should present no particular design difficulty. The complexity of the geometry discouraged a complete analysis. However, using published data on oil-flooded, Zimmern low-pressure compressors, and these data in the figures of appendix B, indicate that an E_m of about 90% can be expected at 200 atm discharge.

Calculations similar to those shown in appendix B indicate that an E_v of about 72% can be expected from a compressor with a 5-cm (2-inch)-diameter rotor operating at design conditions.

The high pressure does not present a great difficulty, but the small size and tight clearances (0.004 mm (0.0015 inch)) will require an advancement of the design and fabrication technology.

[1] Superscripts refer to similarly numbered entries in the Technical References at the end of the text.

The selection of rotor and gaterotor materials suitable for rubbing contact in a water environment might also pose a problem. Rubbing velocities will be as high as 10 m/s (1900 f/m), but rubbing pressures should be below 0.35 kPa (50 psi). The contact pressure between the gaterotors and rotor is low because it is not air-pressure-actuated but is only a result of friction in the bearings which support the gaterotors. This is a demanding requirement but probably within the capability of existing bearing materials. If unacceptable wear is encountered on these surfaces, the nonreversing load and generous sections of the rotors should make it fairly easy to incorporate compliant "floating" seals into these surfaces which use swing-pad bearing technology.[2,3] As shown in figure 2, swing-pad bearings/seals form a hydrodynamic film between the rubbing surfaces which minimizes wear and friction while also controlling leakage. Additionally, the use of water-lubricated swing-pad bearings, to support and position the shafts, would provide a very simple, and therefore, reliable machine with only four rotating parts: one rotor, one inlet shaft seal, and two gaterotors. The relatively light bearing loads, the almost unlimited space available for bearings, and the flexibility of the swing-pad bearing reduce the technical risk involved in such a design.

If conventional oil-lubricated bearings must be used, the Zimmern design is still feasible but would require a longer rotor shaft and 2 to 7 additional shaft seals to isolate the lubricated bearings from the oil-free screw-rotor components. However, such an ROH would still represent a significant improvement over the reciprocating-piston compressors, even though the additional parts complicate a simple design, as well as increase cost, when compared to the water-lubricated design. The development of a shaft seal for discharge pressure would also be required.

WANKEL MECHANISM

The Wankel engine[4] shown in figure 3 is the most highly developed rotary engine concept. There are many design options with this mechanism, some of which are described in Wankel's original work.[5] In its commercially popular form it consists basically of a triangular-shaped (epitrochoidal) rotor which is mounted, via a bearing, on the input shaft with a throw of about one-seventh of the nominal rotor radius (R_i). A stationary gear (radius $2R_i/7$) meshes with a gear (radius $3R_i/7$) attached to the rotor, causing the rotor to turn at one-third input shaft speed. The housing geometry is then chosen to match the path of the three (sealing) points on the rotor. Unlike conventional piston rings which theoretically contact the cylinder along their full outside surface, the motion of the Wankel rotor is such that

the apex (tip) seals must be radiused to maintain only line contact with the housing. Intake and discharge porting is located in the housing. It should be noted that the motion of the rotor is not simple rotation around its own center of gravity but is an eccentric motion which must be balanced. However, the geometry does allow easy and accurate balancing.

The geometry of the compression pockets of a Wankel engine is similar to that of a sliding-vane compressor, but the Wankel design eliminates two of the vane compressor's biggest problems: friction between the vanes and rotor, and the high stress level in the vanes. This allows the Wankel mechanism to operate at higher pressures and to have a higher capacity/size ratio than the vane mechanism. The small number of pockets and the geometry of the apex seals does not encourage high volumetric efficiency, but the high capacity/size ratio would allow a 7-cm (2.7-inch) diameter, 3-cm (1.4-inch) thick rotor to deliver the required capacity with an E_V of 65% and an E_m of about 80%.

These efficiency levels look very encouraging, but there is a fundamental problem with this configuration in that the diameter of the gears and bearings is limited to a small fraction of the rotor diameter. This is not a serious problem with the operating pressures of an internal combustion engine (less than 40 atm (600 psi)), but it effectively prevents the present design from operating as a compressor at the required 200 atm discharge pressure. Furthermore, the noncircular path of the rotor prevents making the design symmetric to balance the pressure forces on the rotor, and it is doubtful if the design could be modified to accommodate the forces imposed by high operating pressures. The original mechanism of Wankel[4,5] has more flexibility because the housing is driven, not the rotor. Therefore, the gears transmit no torque, and there is more room for bearings. It has the added advantage that the uniform rotary motion (of housing and rotor) has no dynamic forces associated with it. Unfortunately, this design is mechanically complex and has very difficult sealing and porting problems. Also, it is questionable if even this version of the Wankel can be fitted with bearings of sufficient capacity for the intended compressor service. Finally, in either configuration, preventing bearing lubricant from entering the oil-free compression area appears to be very difficult. Water-lubricated bearings would eliminate this problem, but the high loads present a formidable problem when compared with those of the Zimmern design. The requirement to lubricate gears further complicates the problem. If sufficient effort was addressed to these problem areas, the Wankel design, with driven housing, could probably be developed to the desired service, but it is not considered an attractive candidate at this time.

CLARKE MECHANISM

The mechanism described by Clarke[6] is a relatively simple mechanism, but its motion is rather difficult to visualize because its rotor moves in three dimensions as it is driven by an angled drive shaft. The single rotor, figure 4, is shaped somewhat like a disk and forms pockets on both sides as it precesses around the central axis of the compressor. Proper synchronization of the rotating and nutating components of this motion allows the sealing contact between the rotor and housing to occur at fixed lines on the rotor.

The operating characteristics of the mechanism undoubtedly depend on the angle of nutation and relative size of the unused central hub. For purposes of this investigation the two-cycle engine described[6] was reconfigured to function as a compressor, and the unit was scaled to the required capacity while maintaining the same relative dimensions.

Calculations indicate that a mechanism with a 4-cm (1.7 inch) diameter rotor could have E_V of 46% and an E_m of 55%. The low mechanical efficiency is primarily due to unbalanced pressure forces caused by the pockets on one side of the rotor being 90° out of phase with those on the other side. The only way to improve this situation would be to couple two units mechanically to balance these forces. The resulting mechanical efficiency would be over 90%. Unfortunately, the compactness and complexity of the geometry would make it difficult to utilize the "floating" seals shown in figure 4-B of appendix B and probably makes the assumed clearance seal rather optimistic; so the E_V appears to be limited to well under 50%. Additionally, the complexity of the design makes it unattractive in the small sizes involved in this investigation.

SCROLL COMPRESSOR

The scroll mechanism[7] shown in figure 5 consists of a stationary "housing" and a orbiting, but not rotating, "rotor" which is driven by an eccentric crane shaft similar to that of the Wankel. Since both the housing and rotor are in the shape of involutes, the proper choice of geometry causes the rotor and housing to effect radial contact at several points, see figure 5. At the start of compression, gas is trapped between two adjacent contact points near the outside of the scroll. As the input shaft rotates, the contact points move inward and the pocket volume is decreased. When the desired pressure is achieved, delivery is effected by eliminating the inside contact point.

The scroll mechanism is currently under development by the A. D. Little Company. A prototype oil-lubricated 16 atm helium compressor has demonstrated an E_V greater than 95%. The high efficiency of the scroll mechanism is a result of the absence of head clearance, the favorable flow characteristics, and the fact that the compression pockets shield the suction and discharge areas from each other. As a result no moving seal is required to seal the full pressure differential across a stage of compression.

Since a lubricated scroll compressor, such as the one mentioned above, uses conventional bearings and sealing technology, there is no reason why it could not be developed into a reliable compressor with a long service life. Unfortunately, a successful oil-free high-pressure air compressor may not be possible for the following reasons:

- The high seal length-to-volume ratio makes contacting seals mandatory to control leakage.

- The operating pressures are too high to obtain acceptable wear life from seal strips made of existing self-lubricating materials.

- The geometry prevents the use of the advanced bearing/sealing technology[a,3] shown in figure 2 and figure 4-B of appendix B.

- The demands which would be placed on the "housing" and "rotor" materials exceed the capabilities of existing self-lubricating materials.

It is conceivable that these problems could be solved by developing bearing and seal designs which use water as a lubricant, but suitable materials and designs are not available at this time.

The drive mechanism of a scroll compressor is very similar to that of a Wankel (driven through the rotor) and can be expected to have a comparable E_m. However, a high-pressure scroll mechanism generates a large thrust force which must be supported and would be the source of additional mechanical losses.

The structural limitation of the scroll mechanism appears to be the deflection of the rotor and housing. By using a small axial dimension and generous cross sections, pressures well in excess of 300 atm could be handled structurally at the expense of capacity and efficiency.

DISK PUMP/COMPRESSOR

The disk turbine was patented by Tesla in 1913, but was only recently (in the 1960's) seriously explored by Rice and others at Arizona State University.[8] The disk compressor, as shown in figure 6, consists of a number of flat, evenly spaced disks mounted on a common central shaft. Unlike conventional dynamic machinery, the disk compressor uses shear force to transfer energy (pressure) to the fluid. The design has received considerable attention because the simplicity of its geometry permits it to be fabricated from virtually any material including ceramics. Additionally, being a device characterized by low specific speed it is cavitation resistant as a pump and offers many of the advantages of a centrifugal compressor for low flow-rate compressor applications. It can accommodate fluids of widely differing properties simply by adjusting disk spacings to maximize the energy transfer with the product fluid. Since the design is axisymmetric there is theoretically no upper limit on its pressure capability, but obviously it would require an increasing number of stages to reach the higher pressure. As an air compressor, the disk mechanism has the disadvantage that losses, which are often size related, become more important with a disk compressor than a centrifugal compressor of the same physical size because the disk compressor has a much smaller capacity. Additionally, the geometry at the discharge is considered very unfavorable by current design criteria for diffusing the high velocity flows without very high losses. Since about half of the generated pressure is in the form of dynamic pressure, this is a critical point.

No information is available on its performance as high or even intermediate pressure compressor. Analysis and experiments at DTNSRDC, however, indicate that in order to use a disk compressor at near optimum conditions would require developing techniques for maintaining a uniform 0.004 mm (0.001 inch) spacing between disks, minimizing the losses mentioned above, and efficiently driving the compressor at 20-30,000 r/m. As an example of the high losses that must be overcome, the disk friction between the last disk and stationary housing can drain as much power as is delivered to the product air. With sufficient effort these problems can probably be diminished but undoubtedly at the expense of simplicity and reliability.

The results of experiments run with a disk compressor at even 10 atm would shed considerable light on the magnitude of these problems, but unfortunately even these tests are not simple. They are comparable to those with centrifugal compressors and require considerable effort and sophistication.

If there were no other means of obtaining the required air, the disk compressor would warrant a more thorough analysis because nothing has been found which prevents it from being modified to the required service. Since the Zimmern requires a much smaller effort, however, and does not require the high speed drive, there seems to be little advantage in developing the disk mechanism as an air compressor.

CENTRIFUGAL COMPRESSOR

During the last 10 years there has been a large effort applied toward developing small-capacity, high-pressure ratio centrifugal compressors for high-performance aircraft and for gas turbine drives for automobiles and trucks. Even these "small" compressors have a capacity 5 times that required by the Navy and deliver air at pressures of less than 15 atmosphere. It is well beyond the scope of this work to examine the feasibility of a very small capacity 200-300 atm compressor. Previous Navy-funded studies[9,10] have proposed multistage designs operating in excess of 100,000 r/m. These high speeds in themselves pose severe bearing and driver problems and would result in a compressor which is bigger, less efficient, and would have less operational flexibility than a positive displacement machine like the Zimmern. The same comment must be applied to the centrifugal compressor as to the disk and scroll compressor; development is possible for the required service, but the effort does not appear warranted in light of the more attractive Zimmern compressor.

LYSHOLM TWIN-SCREW COMPRESSOR

The twin-screw compressor[11] was conceived 40 years ago by Lysholm and has been developed primarily by Svenska Rotor Maskiner, AB, Sweden. Since the more recent development of the asymmetric rotor profile, which improved overall efficiency by as much as 15%, it has become the most versatile and popular positive displacement rotary compressor in the industry.

The geometry of the male and female rotor, shown in figure 7, is such that compression pockets are formed between the rotors and the housing. The female rotor acts somewhat analogous to a moving cylinder, and the male rotor acts like a circular piston which rolls along the cylinder confining and compressing a moving pocket of air. These compressors have typically been oil injected to lubricate rotor-to-rotor contact and absorb compression heat, with delivered pressures to approximately 10 atmosphere. For oil-free service, rotor-to-rotor contact is avoided by external timing gears on the rotor shafts and the compression process is either dry, requiring two stages to achieve 10 atm in order to avoid excessively high discharge temperatures, or in one stage

using water injection to seal the clearances between the rotors and absorb the heat of compression. The "dry" machines, usually in sizes larger than about 100 hp, operate at speeds as high as 12,000 r/m to minimize the effect of leakage. With either oil-free design, nine shaft seals are required to seal the compression areas and confine the lubricating oil to the bearing and gear areas. Multistage lubricated twin-screw compressors have been used for pressures up to 35 atmosphere. Water-flooded helical screw compressors on FFG-7 class ships will deliver 9.5 atm (140 psi), 170 inlet m^3/hr (100 scfm) oil-free air.

In the twin-screw design, the geometry of the compression pocket causes the rotors to be subjected to high axial forces which could be balanced by mechanically coupling two machines and radial forces which cannot be counterbalanced. The radial forces cause the rotors to deflect and separate and impose high loads on the bearings. Unfortunately, the geometry restricts the bearing diameter to about 75% of the rotor diameter. The magnitude of the pressure forces could be reduced by using an 8-lobe male rotor and 12-lobe female rotor configuration instead of the conventional 4-6 lobe design to reduce the area of the rotor exposed to the pressure. Calculations indicate that a compressor with 15-cm (6-inch)-diameter, 8-cm (3.3-inch)-long rotors of the above profile would deliver the required air with an E_V of about 60%. However, even though rotor stress would be low, rotor deflection would be high (0.01 mm) and bearing load would be excessive. Even if the thrust force is supported by hydrodynamic bearings, the maximum estimated bearing life (B-10 rating) would be less than 500 hours. The above rotor design would allow additional support bearings to be placed within the central core of the rotors, but it is highly unlikely that acceptable bearing life could be achieved. The high bearing loads also produce a low E_m (50%), and the requirement to have oil-lubricated timing gears makes the design even less attractive.

OTHER MULTIROTOR DESIGNS

There are numerous other multirotor mechanisms proposed by Wankel[8] and others. The Roots blower and gear pump are the most popular mechanisms in this group. The Unsing engine[11] and the Dean mechanism[12] also fall in this class. However, they are all two-dimensional designs (like the vane) and are all plagued by the common problem of having a long seal length which must separate discharge pressure from the suction pressure. Calculations based on the Unsing engine indicate that when operating at design conditions, with the same clearance seals assumed throughout this report, almost its entire capacity would be consumed by internal leakage. Other mechanisms of this type are expected to have similar performance. Unfortunately, the

requirement that all points on the rotor must act as a seal at some part of the cycle virtually eliminates the possibility of using the "floating" seal already discussed.[2,3] Attempts to run rotor-to-rotor clearances less than those assumed herein would require a complicated and precisely fabricated rotor shape which would compensate for rotor deflection and a very precise rotor positioning system.

The E_m of the multirotor designs was not calculated because their low volumetric efficiency eliminated them as viable candidates. It can be noted however, that many of these designs would also suffer the effects of high unbalanced forces which plague the Lysholm compressor.

OTHER SINGLE-ROTOR MECHANISMS

Sliding-Vane Compressor

The original rotary mechanism, and the simplest, is the rotary sliding-vane compressor shown in figure 8. Commercially it is used for pressures up to about 20 atmosphere. Its operating characteristics and potential as a ROH are discussed in some detail in appendix B. In operation, vanes mounted in slots in the rotor move radially to seal the varying distance between the rotor and housing. The relatively high stress levels in the vanes, and the sliding contact between vanes and rotor, are the limiting structural items. Theoretically they could be handled by using thicker vanes and limiting the movement of the vanes, but this is accomplished at the expense of a considerable loss in efficiency. Practically, the design is limited to the pressure range in which it is now used.

It is calculated in appendix B that a vane compressor with an 8-cm (3.3-inch) diameter rotor could deliver the required air with an E_v of 50% and an E_m of 56% if mechanical problems discussed in appendix B could be solved. The mechanical efficiency of a vane compressor could be improved by using symmetric discharge, but new seal technology would still have to be developed to seal the varying space between the rotor and housing before it could be used at pressures above 20 atmosphere.

Liquid-Ring Compressor

The liquid-ring or liquid-liner compressor shown in figure 9 is used for vacuum service, and compressor service up to 10 atm, often with corrosive gases. As with the vane compressor, it uses a single rotor which rotates in a nonconcentric housing. The spinning of the rotor, with its integral blades, throws sealant

(usually water) outward to the housing. This ring of water seals the varying radial gap between the rotor blades and the housing and acts like a piston moving radially between the vanes. This design relies on the inertia and centrifugal force of the water to prevent the water ring from being "collapsed" or water being pushed back to suction by the pressure differential across the machine. The pressure capability is a function of rotational speed. The large hydraulic losses associated with higher speeds limit its utility to about 15 atmosphere.

Other Designs

The Renault-Rambler[13] and Sarich engines[14] were not considered viable candidates because they require cylinder valving and mechanical complexity and present considerable sealing and bearing problems when operated as an oil-free compressor.

The "cat and mouse" or scissors engines are so named because pairs of alternating pistons run away from and catch up with each other as they travel around a toroidal chamber. The Tschudi, Kauertz, and the Maier engines[13] are representative of this class of mechanisms. The two pairs of toroidal pistons are externally controlled by cams and gears so that the space between pistons increase as they pass the suction port and decreases as the discharge port is approached. This design imposes no radial or thrust forces on the drive mechanism, but the cylinder pressure forces and the acceleration forces associated with the nonuniform velocity of the pistons imposes a very high torque loading on the cam and gear system. The geometry does not limit the size of these members; so it is conceivable that the high loads could be accommodated, but it would undoubtedly be a low efficiency, low reliability design. Additionally, the large seal area which must separate the suction and discharge pressures gives a volumetric efficiency of only 20% at design conditions. Chinitz[13] also discusses a group of designs which employ reciprocating pistons moving in a rotating cylinder housing. They were not seriously examined because they all appear to be mathematical curiosities which, if fabricated, would be volumetrically inefficient and mechanically complex.

DISCUSSION

GENERAL

The reported results indicate that the Zimmern single-screw compressor is the only design which could be converted to oil-free, 200 atm, low-capacity service through the application of

available bearing and seal technology. The following discussion therefore addresses almost exclusively this design.

Successful operation of a ROH compressor requires the solution of three interdependent problems, preventing oil contamination of the product air, supporting the pressure forces, and efficiently compressing (sealing) the air. The proposed Zimmern design is attractive because it avoids two of these problems rather than solving them. The first problem is avoided by the complete elimination of oil from the machine and the second is greatly simplified by the symmetry of the geometry. Efficiently compressing the air will still require considerable development effort, but even this is much simpler than with most of the other designs considered.

Compared to a reciprocating piston compressor, the Zimmern ROH would offer the following advantages:

- Size and weight will be reduced by 50%.
- Initial compressor cost will be reduced at least 30%.
- Reliability will be improved 400%.
- Number of parts will be reduced by 75%.
- Multiple compression stages can be mechanically independent which offers more system flexibility.
- Low discharge temperatures (49° C (120° F)) allow greater flexibility in system design and greater safety.
- Smallness of the compressor allows for modular replacement, while the simplicity of the design makes shipboard overhaul feasible.
- Pure rotary motion promotes very low vibration in a machine which should meet MIL-STD 740B requirements for type II equipment.

APPLICATION

The study has intentionally kept open various options as to how such ROH technology could or should be applied. Those options envisioned include following conventional shipboard practice and designing the ROH to compress air from 0-200 atm (3000 psi) in as few as three stages. Used as a booster compressor, it could take suction from ships' low-pressure air system or gas turbine bleed

air and deliver at 200 atm (3000 psi). Another alternative is to power each stage individually so that each can operate independently or all at once. Regardless, the small number of moving parts, coupled with their individually small size (5 kg maximum), should minimize the support function and increase reliability. As with the twin-screw compressor, the smallness of each stage of compression permits modular replacement with major repairs conducted at a service facility. However, the absence of precision timing gears and the reduction in the number of shaft seals makes shipboard repair feasible, if required.

The probable need to have only three stages of compression, and the fact that they can be mechanically independent, allows a highly flexible shipboard air system. The same hardware including castings could be used, at different times, as a low-pressure compressor (up to about 12 atm) or as the lower stage of compression of the high-pressure rotary compressor. Similarly, there may be advantages in some cases in the use of the first two stages of compression to deliver air up to 100 atm (1500 psi) when air is required at pressures between 13-100 atm rather than reducing it from 200 atm as is the current practice.

AIR SEALS

All calculations were based on using noncontacting seals with clearances in the range of 0.004 mm (0.0015 inch) and neglecting the effects of wear and nonpressure-actuated deformation. This may be optimistic, but the generous cross-section of the Zimmern gaterotors allows the use, if required, of compliant seals mounted in these surfaces. Bearings[4] such as those shown in figure 2 could be used as seals which would minimize any wear problem at these surfaces.

VALIDITY OF THE ANALYSIS

Many approximations, particularly of geometry, were necessitated by the scope of this work. It has already been estimated in appendix A that the leakage calculations may contain as much as 10% error and undoubtedly the other approximations involved additional error. However, regardless of the error in the approximation, the relative ranking of compressors should be unaffected.

To determine the validity of these approximations, the volumetric efficiency of a low-pressure Zimmern and Lysholm compressor, and the mechanical efficiency of a Clarke compressor (engine), were calculated by the same procedures and approximations as used within. The calculated values (90%, 80%, and 65%) compared very well with the reported values (93%, 85%, and 75%).

Obviously, this does not establish any sort of accuracy, but it does at least show that the conclusion based on these calculations are reasonable.

CONCLUSION

The Zimmern compressor is the rotary mechanism which can be developed into a technically attractive ROH compressor to meet the Navy's needs. Conversion of the Zimmern compressor to this service will employ advanced seal, bearing, and material technology but should not require exotic or complicated designs.

RECOMMENDATIONS

Based on the above results, it is recommended that a low-pressure Zimmern compressor be purchased, converted to oil-free service, and used to develop much of the technology required for a high-pressure unit. It is also recommended that design of a high-pressure unit be initiated quickly so that the developmental work with the low-pressure unit can be guided by the needs of this design.

TECHNICAL REFERENCES

1 - Zimmern, B., and G. Patel, "The Design and Operating Characteristics of the Zimmern Single-Screw Compressor," from Proceedings of Purdue Compressor Technology Conference (1972)

2 - Thomas, David, "High-Performance Vane Pump for Aircraft Hydraulic Systems," AFAPL Rept TR-75-94 (Oct 1975)

3 - Greene, J., "The Swing-Pad Bearing - A New Concept in Sliding Surface Bearings," DTNSRDC Rept 76-0055 (Dec 1976)

4 - Ansdale, R. F., "The Wankel RC Engine, Design and Performance," published by ILIFFE Books Limited (1968)

5 - Wankel, F., "Rotary Piston Machines," published by ILIFFE Books Limited (1965)

6 - Clarke, J. M., <u>et al.,</u> "A New Class of Rotary Piston Machines Suitable for Compressors, Pumps, and Internal Combustion Engines," published in Proceedings of the Institution of Mechanical Engineers (V186) (1972)

7 - Moore, R., <u>et al.</u>, "A Scroll Compressor for Shipboard Helium Liquifier Systems," presented at Purdue Compressor Conference (1976)

8 - Rice, Warren, "An Analytical and Experimental Investigation of Multiple Disk Pumps and Compressors," ASME Paper 62-WA-191 (Nov 1962)

9 - Smith, L. F., "Feasibility of a High-Pressure Centrifugal Air-Compressor for Submarine," MTI, Inc., under Navy Contract NObs 92476 (1967)

10 - Willenbrock, A., Jr., "Feasibility of a High-Speed Centrifugal Compressor for High-Pressure Air Service Aboard Submarines," Navy Contract NObs 90393 (1965)
11 - Taft, G., "Selection and Application of Industrial Screw Compressors," published in Proceedings of Purdue Compressor Technology Conference (1972)
12 - Dean, W. C., "A New Rotary Piston Engine," published in Mechanical Engineering (Oct 1974)
13 - Chinitz, Walter, "Rotary Engines," published in Scientific American (Feb 1969)
14 - Ananymous, "The Sarich Orbital Engine," published in Marine Engineers Review (Dec 1973)

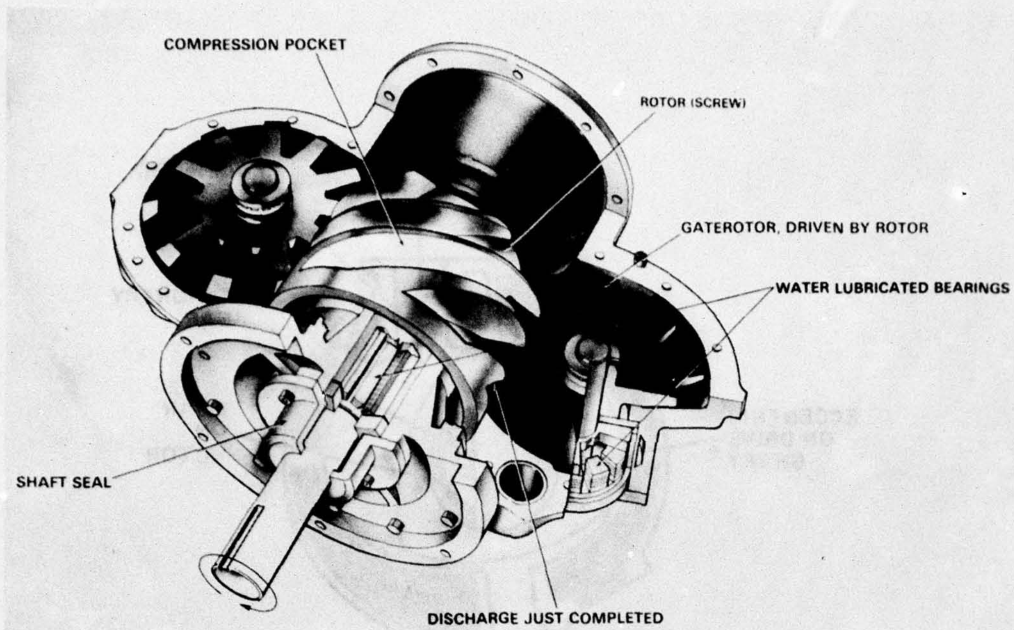

Figure 1
Oil-Free Single-Screw Compressor

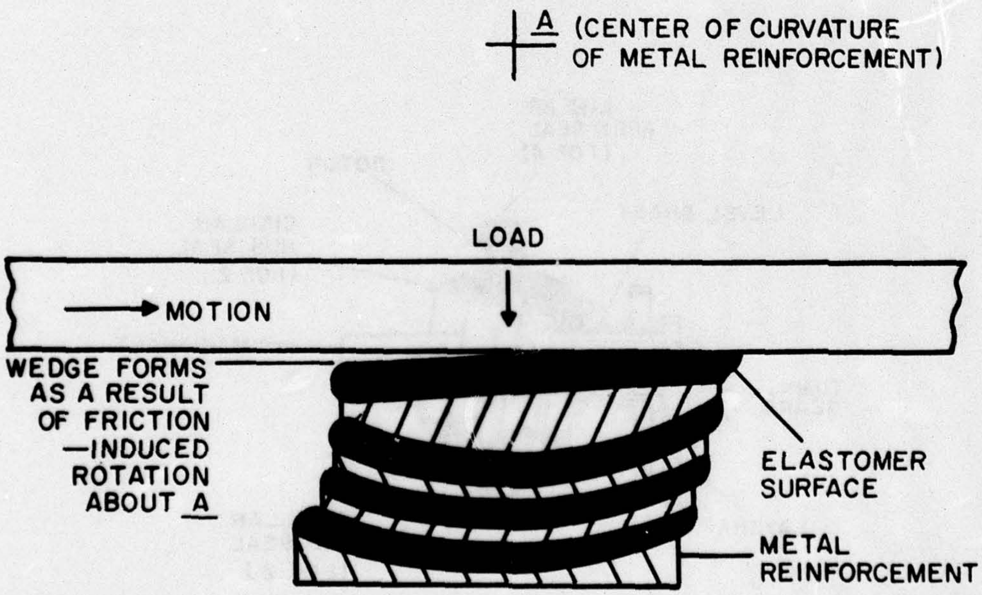

Figure 2
Swing-Pad Bearing

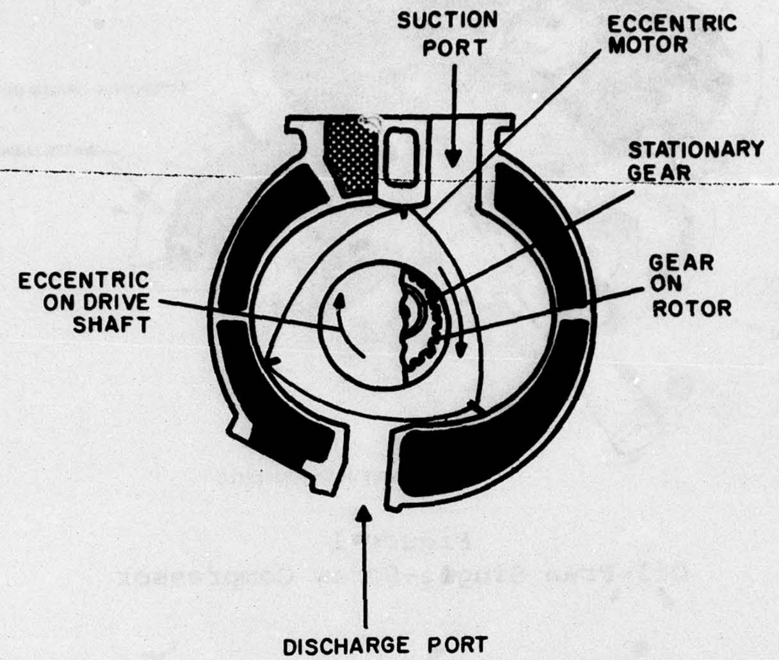

Figure 3
Wankel Mechanism

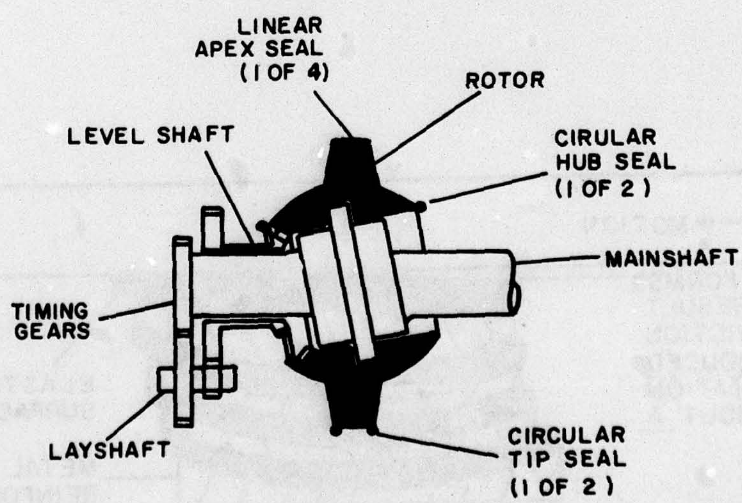

Figure 4
Clarke Mechanism

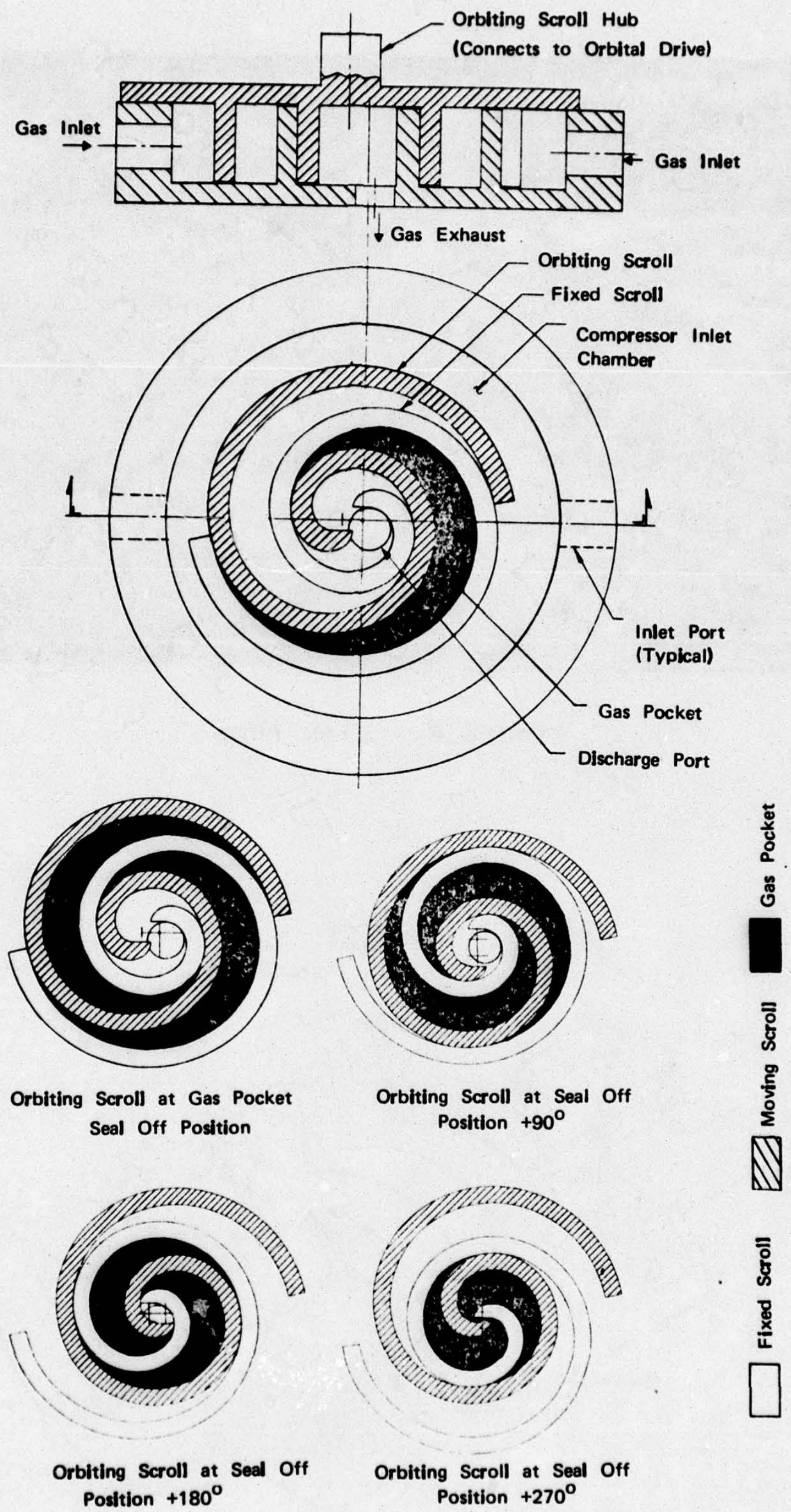

Figure 5
Scroll Compressor

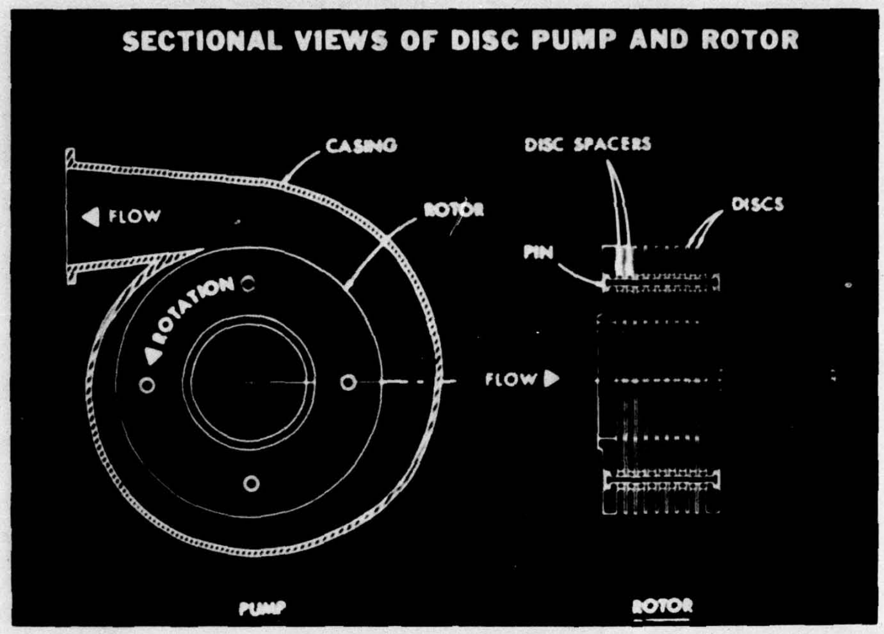

Figure 6 - Disk Pump

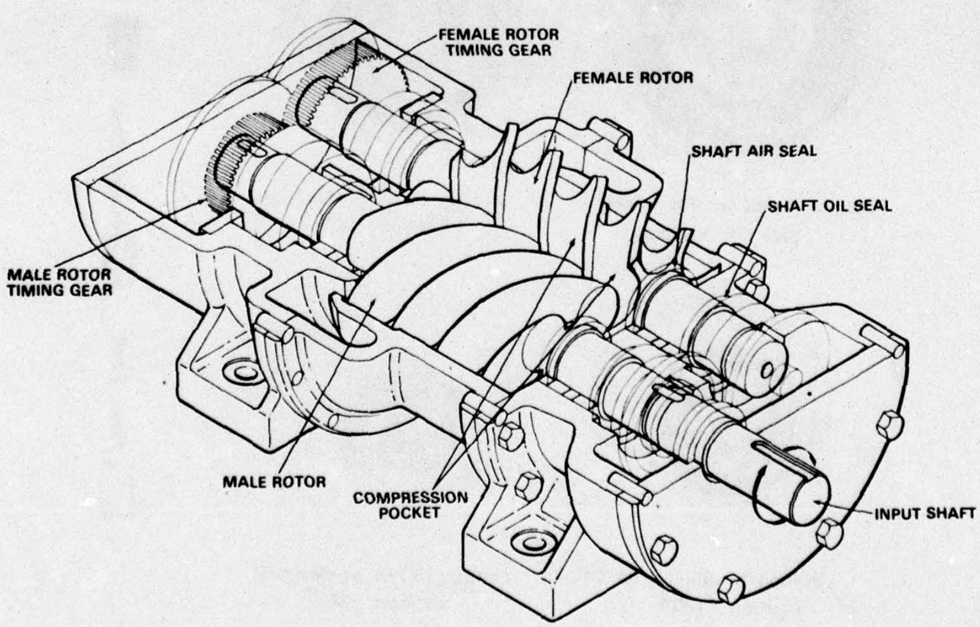

Figure 7
Twin-Screw Compressor

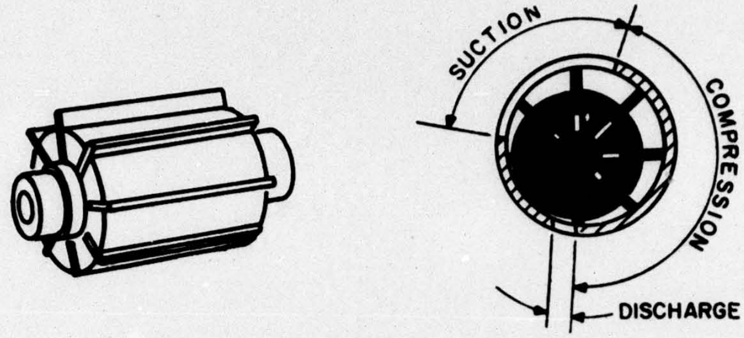

Figure 8
Vane Compressor

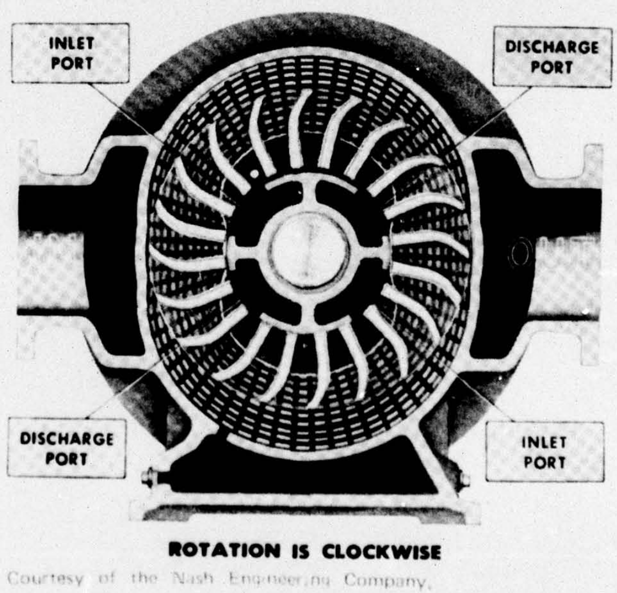

Figure 9
Water Ring Compressor

APPENDIX A
CALCULATION OF SEAL LEAKAGE

References:

(a) Shapiro, Ascher H., "The Dynamics and Thermodynamics of Compressible Fluid Flow," Ronald Press (1953)

(b) Zuk, John, et.al., "Compressible Flow Across Shaft Face Seals," Paper H6 at Fifth Annual Fluid Sealing Conference, University of Warwick, Coventry, England (Apr 1971)

(c) Boyce, M. P., and A. R. Desai, "A Theoretical Analysis of Nonisentropic Flow of a Compressible Viscous Gas in Narrow Passages," MTI 67TR8 (10 Feb 1967)

All seals in the text were assumed to be thin ducts, of constant cross-section, with a ratio of length to hydraulic radius of 25. Leakage rates were calculated for dry air and stationary walls. The calculated rates were then modified to show the influence of moving walls and liquid water. In effect the final calculations assume the volumetric leakage through the seals is all liquid. The effect of this assumption is discussed at the end of this section. This procedure was used, rather than direct calculation of water leakage under the same condition, because it allowed the effect of the cooling water to be determined and the error within this assumption to be estimated. The mass flow rate becomes:

$$\frac{XX'X'' \; AP}{(RG\;T)^{1/2}}$$

where X, X′, X″ indicate the effect of compressibility, moving wall, and water, respectively, on the leakage rate.

LEAKAGE OF DRY AIR

Calculations based on several procedures in the literature were performed to estimate the flow through a thin channel. These indicate that the classical theory of Shapiro, reference (a), tends to overestimate the leakage by as much as 20% when compared to the experimentally verified theory of Zuk and others, reference (b). Since the calculations required by Shapiro's theory are much simpler, his theory was used throughout. The results of this theory which assumes isentropic entrance, adiabatic flow, and perfect gas are shown in figure 1-A. Polynomial approximations of these results were used to allow closed form solutions to be obtained.

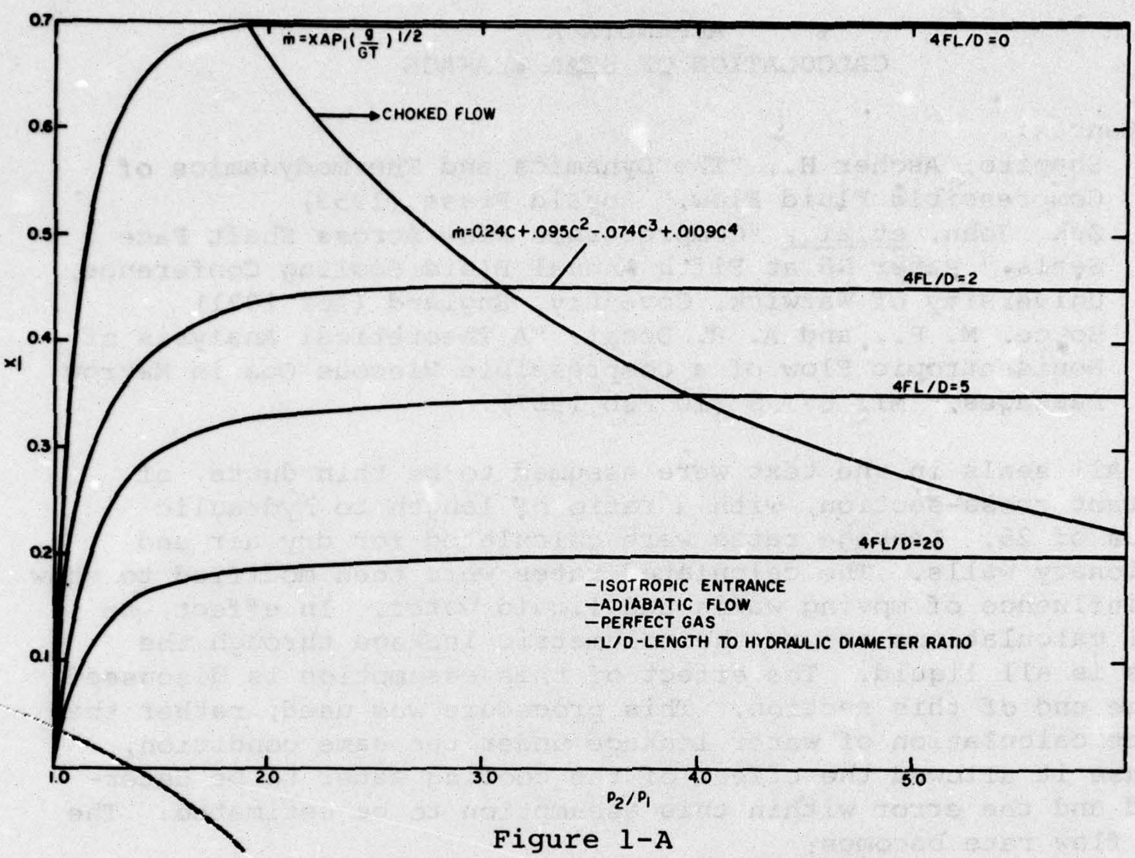

Figure 1-A
Leakage of Dry Air Through Narrow Duct

THE EFFECT OF A MOVING WALL

The effect of wall motion perpendicular to the direction of flow has been widely studied because of its practical importance in noncontacting, rotating shaft seals. The results of Zuk, reference (b), however, indicate that the effect of this type of motion is relatively unimportant for this application. The effect of wall motion parallel to the direction of flow motion was studied by Boyce and Desai, reference (c). They found that for a similar geometry, the flow rate increased about 23% for each 100 m/s of forward wall motion with the other wall remaining stationary. Wall motion in the other direction, of course, reduces the flow rate an equivalent amount. Therefore, if the velocity of one wall is Rw, the effect of the moving wall on the leakage rate is:

$$X^1 = 1 \pm 0.234 \frac{\text{wall velocity}}{100 \text{ m/s}} \quad \text{(sign depends on direction of wall motion)}$$

$$= 1 \pm 0.17R\left(\frac{w}{120\pi}\right).$$

THE EFFECT OF LIQUID WATER

The presence of more viscous and incompressible water generally reduces the volumetric flow rate. Since the flow rate of the liquid depends on pressure difference and the flow rate of the air depends on pressure ratio, the amount of reduction is not a simple formula. Leakage under various conditions is shown in the following tabulation.

Pressure Atm		Flow Rate, cm^3/min[1]			Flow Air, Flow Water X''	
		At 49° C		Air[2] at Compression Temperature	At 49° C	Compression Temperature[2]
P_1	P_2	Water	Air			
68	54	167	573	558	0.292	0.300
68	41	250	700	657	0.360	0.382
68	14.5	305	780	645	0.393	0.447
68	9.5	322	780	610	0.414	0.582
71	68	76	304	301	0.252	0.254
75	68	118	422	420	0.272	0.275
81	68	167	560	547	0.300	0.306
102	68	250	680	647	0.370	0.389
123	68	375	737	688	0.509	0.546
137	68	398	760	700	0.524	0.569
200	68	532	780	686	0.718	0.820
200	102	486	737	665	0.641	0.731
200	137	398	680	647	0.586	0.616
200	170	292	560	547	0.504	0.515
200	190	175	348	345	0.502	0.506

[1] Based a 0.25 mm (0.01 inch) long duct with a cross-section of 0.05 x 1 mm (0.002 x 0.039 inch) and a surface roughness of 77×10^{-8} m (30×10^{-6} in.) volume referred to P_2 and 49° C.

[2] Upstream air temperature based on compression with P_1 and 49° C with a polytropic coefficient of 1.3 and volume referred to P_2 and 49° C.

It is evident that water greatly reduces the volume flow rate at low pressures (X'' is small) but has a much lesser effect at higher pressures (X'' is near 1)

DISCUSSION

The cases of all water and no water leakage represent the extreme cases. The actual leakage rate must be between these two limits. The assumption that all leakage occurs as liquid

underestimates the leakage by about 30% compared to a 50% water/ 50% air mixture, but since Shapiro's theory overestimates the leakage by about 20%, the calculated leakage is therefore about 10% low. The result of the higher leakage rate would be a lower E_V. For example, if the additional 10% leakage was included, the E_V would be reduced from 72% to 69%. Since this does not significantly affect the results, it was neglected.

APPENDIX B

SAMPLE CALCULATIONS COVERING ANALYSIS OF VANE COMPRESSOR

References:

(a) Thomas, David, "High-Performance Vane Pump for Aircraft Hydraulic Systems," AFAPL Rept TR-75-94 (Oct 1975)

(b) Greene, J., "The Swing-Pad Bearing - A New Concept in Sliding Surface Bearings," DTNSRDC Rept 76-0055 (Dec 1976)

(c) Chlumsky, V., Reciprocating and Rotary Compressors, Pub. by SNTL of Prague (1965)

CAPACITY AND VOLUMETRIC EFFICIENCY

A schematic representation of a vane compressor, including leakage paths, is shown in figure 1-B. The notation and geometric assumptions are also shown in this figure. The assumptions of isothermal compression, pressure unaffected by leakage, a uniform time rate decrease of pocket volume during the compression process, and the conditions listed in table 1-B give a pressure distribution of:

$$C = \frac{P}{P_1} = \begin{cases} (1-0.318\ \theta)^{-1} & 0<\theta<\frac{2\pi}{3} \\ 3 & \frac{2\pi}{3}<\theta<\frac{7\pi}{6} \\ 1 & \frac{7\pi}{6}<\theta<2\pi \end{cases} \qquad \text{(B-1)}$$

Neglecting inefficiencies of inlet which are discussed later, the capacity of a vane compressor is equal to the product of the density, velocity, and area at the compressor inlet (θ=0). If the vanes are assumed to occupy 20% of the volume the inlet capacity is:

$$m_1 = \left(\frac{P_1}{GT_1}\right) w \frac{\left(R_o + R_i\right)}{2} (2L)\ (R_o-R_i)\ (0.8)$$

$$m_1 = 1.348\ L \left(R_o^2-R_i^2\right)\left(\frac{w}{120\pi}\right) \frac{kg}{min} \qquad \text{(B-2)}$$

The delivered capacity of the machine will be decreased by leakage and air which is not expelled from the pocket at discharge.

Leakage from one pocket to another will cause additional power consumption but will not affect the E_v of the machine. All leakages are calculated according to the procedure described in appendix A.

TABLE 1-B
DESIGN CONDITIONS

Discharge Pressure, atm	204	(3000 psi)
Inlet Pressure, atm	68	(1000 psi)
Inlet Air Temperature, °K	321	(120° F)

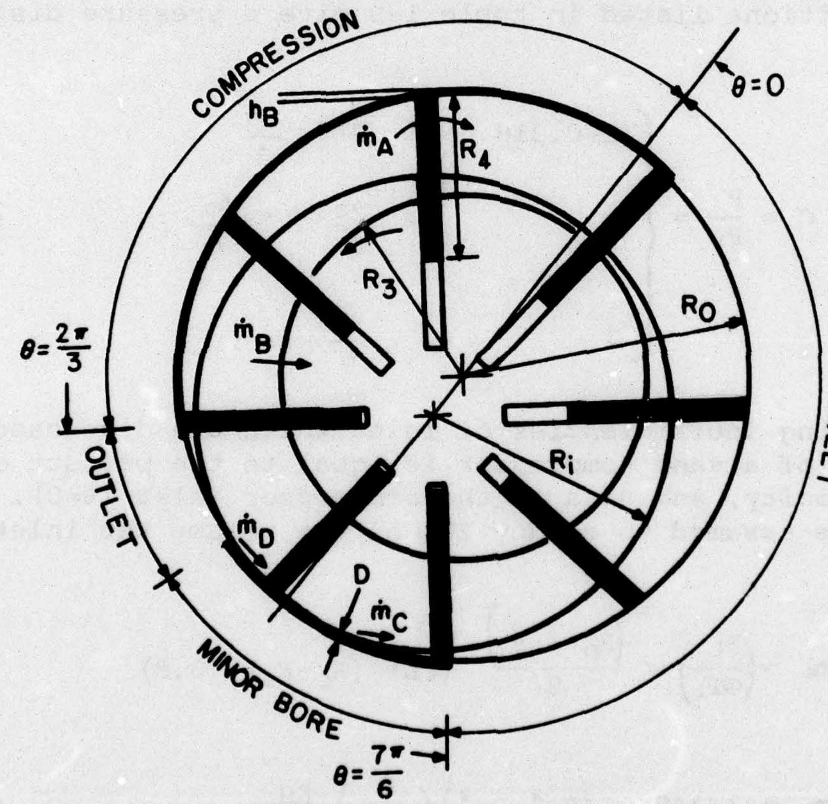

Figure 1-B
Geometry of Vane Compressor

The pressure in the recently closed pocket will rise above 68 atm as the pocket volume decreases. Air will, therefore, leak into the succeeding, unclosed, pocket and displace air which would otherwise have been injested in the suction. For multi-pocket machines such as the vane, this could be reduced by altering the geometry to delay compression until the succeeding pocket is closed, but it is retained in these calculations because it is inherent in many of the designs being considered. The leakage can be calculated as:

$$m_a = \frac{N}{2\pi} \int_0^{\frac{2\pi}{N}} \frac{XX''AX'Pd\theta}{(GT)^{\frac{1}{2}}}$$

$$(X' = 1 - 0.017\ h_o\left(\frac{w}{120\pi}\right)\ ;$$

$$X'' = 0.26\ ;\ X = 0.24C + 0.0954C^2 - 0.0744C^3 +$$

$$0.0109\ C^4)$$

$$A = h_a L + 2\ h_b\ R_4)$$

$$m_a = 11.88\ A\left(1 - 0.017\ R_o\left(\frac{w}{120\pi}\right)\right)(N = 10) \qquad \text{(B-3a)}$$

$$m_a = 11.30\ A\left(1 - 0.017\ R_o\left(\frac{w}{120\pi}\right)\right)(N = 20) \qquad \text{(B-3b)}$$

Product air also can leak radially inward through the gap between the housing and the rotor. By conservatively assuming that the core of the rotor is vented to suction, the leakage rate can be calculated as:

$$m_b = \frac{1}{\pi} \int_0^{\pi} \frac{X'X''AXPd\theta}{(GT)^{\frac{1}{2}}}$$

$$(X' = 1;\ X'' = 0.1 + 0.2C;\ A = \pi R_3\ h_3\ \ ;\ X \text{ - same as above})$$

$$m_b = 224\ R_3\ h_c \qquad \text{(B-4)}$$

Undelivered volume between the rotor and housing is carried back to suction. This represents lost capacity and can be calculated as a simple flow in the same manner as the inlet.

$$m_c = 0.8LDR_i\ w + 2\ R_4 h_d \left(R_i - \frac{R_4}{2}\right) \frac{Pw}{GT}$$

$$m_c = \left(4.046\ DR_i L + 10.11\ R_4 h_d \left(R_i - \frac{R_4}{2}\right)\right) \frac{w}{120\pi} \quad \text{(B-5)}$$

The first term represents air carried between the outside of the rotor and the housing; the second term represents air carried between the side of the rotor and the housing. The reexpansion of the pocket is delayed so the pressure within the recently discharged pocket is kept as high as possible to reduce the leakage from the discharge port into this pocket. However, the small amount of air remaining in the pocket after discharge is lost as leakage (m_b) in about the same time it takes the pocket to leave. If it is assumed that the pressure distribution during this process is similar to that during the compression process the leakage from discharge into this pocket can be calculated in a manner similar to m_a.

$$m_d = 28.15 \left(1 + 0.017\ R_o \frac{w}{120\pi}\right) (Lh_A + 2\ R_4 h_B)\ \text{kg/min} \quad \text{(B-6)}$$

The values listed below were used in subsequent calculations because they were found to represent approximately the optimum practical design

$$R_i/R_o = 0.95$$

$$R_3/R_o = 0.50$$

$$R_4/R_o = 0.50 \quad \text{(B-7)}$$

$$D/R_o = 0.003$$

$$N = 20$$

h_a/R_o = see equation (B-8)

$$h_b/L = 0.005$$

$$h_c = 0.004 \text{ cm}$$

$$h_d/R_o = 0.002\left(\frac{w}{120\pi}\right)^{\frac{1}{2}} \quad \text{(B-7)}$$

The value of h_d was chosen to minimize the total, volumetric, and mechanical losses.

$$\frac{h_a}{R_o} = 0.0005 + 5.42 \times 10^{-7}\left(\frac{L}{R_o}\right)^4 \quad \text{(B-8)}$$

(The last term represents the average rotor defection due to pressure forces.) Equations (B-3) thru (B-8) can be used to calculate the total decrease in capacity $m_L = m_a + m_b + m_c + m_d$

$$m_L = 0.448\, R_o + 0.0394 LR_o + 2.13 \times 10^{-6}\left(\frac{L}{R_o}\right)^4 LR_o + 0.0113\, LR_o^2\, \frac{w}{120\pi} + 1.54 \times 10^{-7}\left(\frac{L}{R_o}\right)^4 LR_o^2 + 0.007\, R_o^3 \left(\frac{w}{120\pi}\right)^{3/2} \quad \text{(B-9)}$$

The delivered capacity of the machine becomes $m_t = m_1 - m_L$

$$m_t = 0.12 LR_o^2\, \frac{w}{120\pi} - \Bigg(0.448\, R_o + 0.0394\, LR_o + 2.13 \times 10^{-6}\left(\frac{L}{R_o}\right)^4 LR_o + 1.54 \times 10^{-7}\left(\frac{L}{R_o}\right)^4 LR_o^2 + 0.007\, R_o^3 \left(\frac{w}{120\pi}\right)^{3/2}\Bigg) \quad \text{(B-10)}$$

Equation (B-10) can be used to calculate the required axial length for various capacities, speeds, and housing diameters. The results are shown in figure 2-B. Equations (B-2), (B-9), and (B-10) can be used to calculate the volumetric efficiency E_v. The results of these equations are given in figure 3-B. It indicates that the use of high rotational speed (18,000 rpm) does not offer a significant increase in E_v when compared to that obtained from 3,600 rpm.

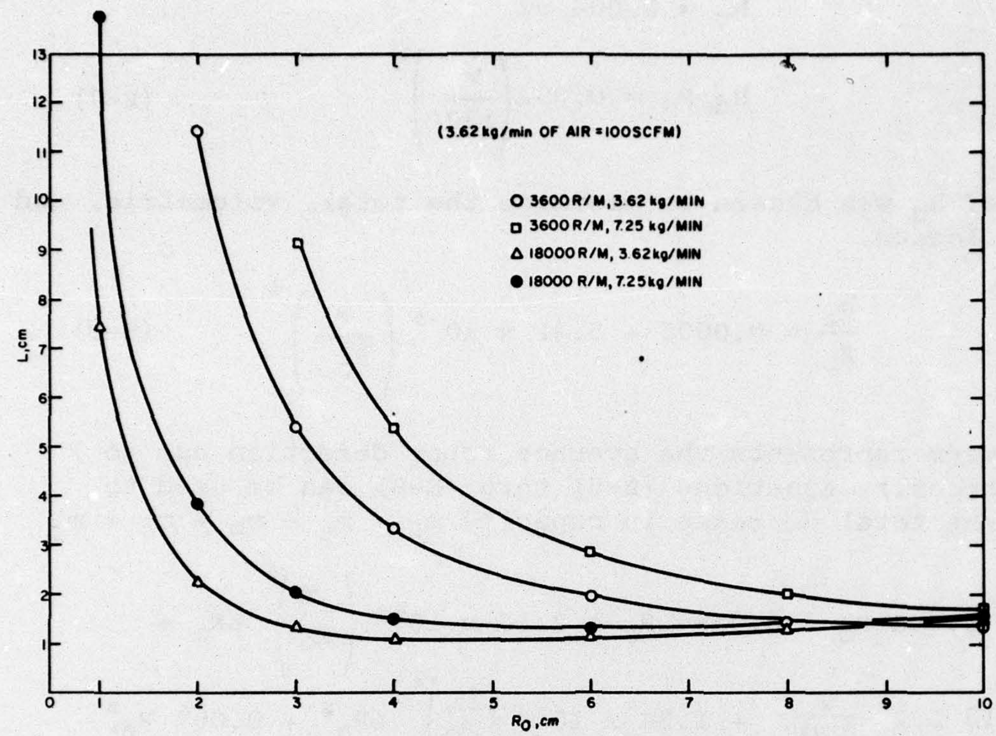

Figure 2-B
Rotor Geometry of Vane Compressor

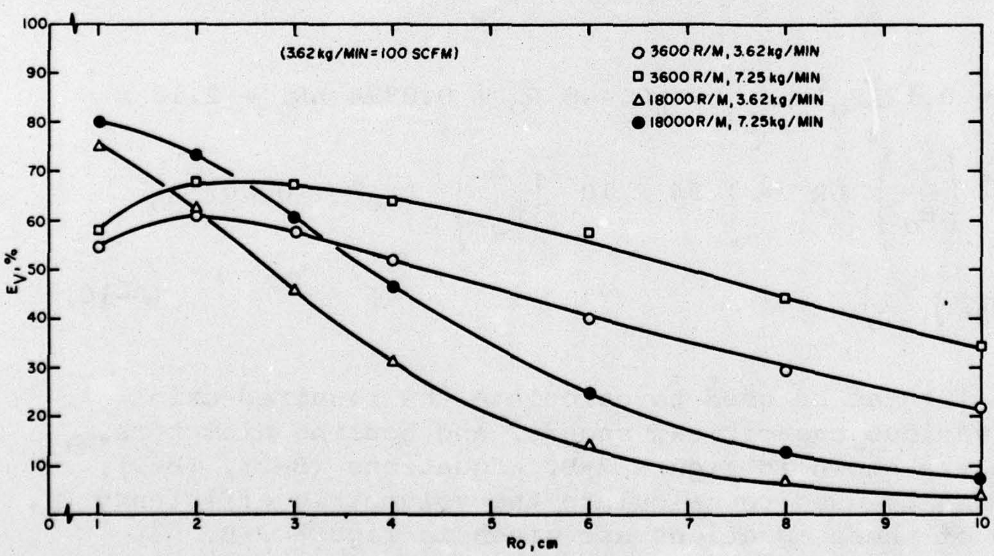

Figure 3-B
Volumetric Efficiency of a Vane Compressor

The high E_V calculated for small rotors at 18,000 rpm is probably not valid because, at these sizes, the assumed clearances may be unrealistic. The lower E_V at larger rotor sizes and high rotational speeds is primarily a result of the need to choose a larger rotor-housing clearance (h_d) to minimize mechanical losses. It might be possible to obtain a slightly higher efficiency at speeds above 3,600 but below 18,000 rpm; this was not immediately investigated. Lower speeds would yield a lower volumetric efficiency.

At the optimum geometry for 3600 rpm, based on overall compressor efficiency, (R_O = 4 cm, L = 3.33 cm) the volumetric efficiency of a 3.6 kg/min compressor is approximately 52% which is about 20% below that of a comparable reciprocating compressor with contacting seals. With this geometry, the various leakages are calculated as:

m_t (delivered capacity) - 3.62 kg/min

m_a (inlet leakage) - 0.14 kg/min

m_b (leakage radially inward) - 1.78 kg/min

m_L (undelivered volume - 1.06 kg/min

m_d (leakage past discharge) - 0.40 kg/min . (B-11)

The geometry could undoubtedly be modified to improve the volumetric efficiency; particularly, m_b could be reduced by using several concentric seals, but this was not examined because E_V is not the limiting factor of a vane compressor. m_c, which can only be reduced at the expense of mechanical losses and m_d are large because the vane rotor reduces the pocket volume without reducing its axial and circumferential dimensions, leaving large leakage paths and a relatively large undelivered "clearance" volume. Zimmern and Lysholm screw compressors reduce all pocket dimensions and have virtually no clearance volume.

Some leakages could be reduced by using hydrodynamic seals (reference (a) as shown in figure 4-B in place of noncontacting clearance seals. Water lubricated bearings (reference (b)) could serve the same function without the complexity. The resulting seal which would have no internal rubbing parts could also be used to support the centrifugal force on the vanes and solve the problem of accurately positioning the vanes.

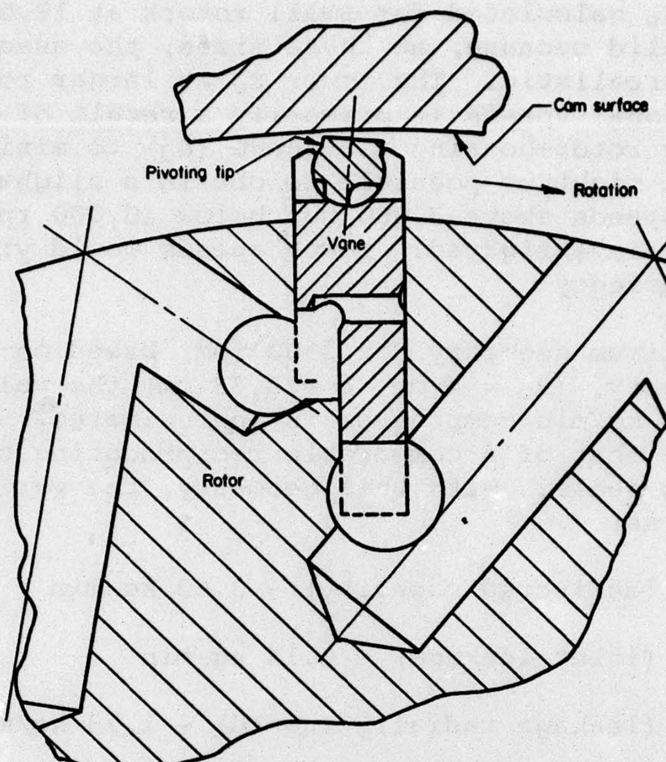

Figure 4-B
Pivoting Tip Seal

MECHANICAL EFFICIENCY

Besides the work put into the compression process, work must be used to overcome frictional losses. The most common loss is that resulting from friction in the bearings supporting the rotor shaft.

$$S_A = f\ (\text{load supported})\ (\text{bearing radius})\ w$$

$$= 0.001\ (544\ LR_O)\ (4\ LR_O)^{1/2} w \qquad \text{(B-12)}$$

$$= 0.08\ (LR_O)^{3/2} \left(\frac{w}{120\pi}\right) \text{kw}$$

A second source of loss is the friction between the vanes and the rotor as the vanes move radially. Exact calculations are tedious, but the losses can be estimated by:

$$S_B = N\ f\ A\ P\ V$$

$$= 20\ (0.15)\ (R_4 L) \left(\frac{P_f - P_1}{4} \right) 2\ (R_o - R_i)\ \frac{w}{120\pi}$$

$$= 0.03\ R_o^2\ L \left(\frac{w}{120\pi} \right)\ \text{kw} \qquad \text{(B-13)}$$

A third type of inefficiency for a positive displacement rotary machine is suction (charging) and discharge losses. This is a result of the necessity to accelerate the product air and move it through the flow passages of the compressor. It is generally approximated by two velocity heads at rotor tip speed for suction losses and one velocity head for discharge losses. Assuming $E_V = 50\%$, this becomes:

$$S_C = 2\ (R_o w)^2\ m_t/gE_V + (R_i w)^2\ m_t/g$$

$$= 0.0021\ R_o^2 \left(\frac{w}{120\pi} \right)^2\ \text{kw} \qquad \text{(B-14)}$$

Losses also occur when two parts with relative motion are separated by a small gap filled with a viscous fluid such as water or air. The power lost can be conservatively estimated by $2uV^2\ A/3h$. The factor 2/3 is used because the gaps are assumed to be much larger during the suction part (1/3) of the cycle. There are three areas worth considering: the space between the side of the rotor and the housing, the space between the side of the vane and the housing, and the space between the tip of the rotor and the housing. The losses occurring in these areas can be represented by:

$$0.8\ (2u/3h_d \int_{R_3}^{R_i} R^3\ w^2\ 2\ \pi dR),\ 0.2(2u/3h_B) \int_{R_3}^{R_o} R^3 w^2 \pi dR),$$

and

$$0.2\ (2u/3h_A)\ \ R_o^3\ w^2\ L\ 2\pi), \qquad \text{(B-15)}$$

respectively. Substituting equations (7) and (8) into equation (15) gives:

$$\text{for air } S_D = \left[5.1 \times 10^{-6}\ R_o^2\ L + 6.53 \times 10^{-7}\ R_o^4 \left(\left(\frac{120\pi}{R_o^2 w} \right)^{1/2} + \frac{1}{L} \right) \right] \left(\frac{w}{120\pi} \right)^2\ \text{kw} \qquad \text{(B-16)}$$

$$\text{for water } S_D = \left[72.5 \times 10^{-6} R_O^2 L + 137 \times 10^{-7} R_O^4 \left(\left(\frac{120\pi}{R_O^2 w} \right)^{1/2} + \frac{1}{L} \right) \right] \left(\frac{w}{120\pi} \right)^2 \text{ kw} \quad \text{(B-16a)}$$

The results of equations (B-12) through (B-16) are given in figures 5-B and 6-B. With the optimum conditions (3600 rpm, R_O = 4 cm) the mechanical efficiency is about 56%. Figures 5-B and 6-B also show that the bearing and vane frictions are the largest losses. These are also experienced by commercial vane compressors but do not represent such a penalty because a low-pressure compressor can be designed with a much higher capacity than a high-pressure machine of the same dimensions and will also have a higher E_V. Since these differences in design do not greatly affect losses, the losses experienced with a low-pressure machine represent a much smaller fraction of the air power delivered. Bearing friction and rotor deflection could be almost eliminated by using a symmetric discharge, but this would also mean a large machine would be required to deliver the same capacity and would double m_C and m_D from equation (B-12). Vane friction could be reduced by pressure balancing of the vanes but probably not to an acceptable level. The other losses discussed are not pressure related and are unavoidable unless free spinning side disks and housing liners (reference (c)) are employed. Even with these complex modifications it is doubtful if much improvement can be obtained. It is fortunate that these losses are relatively unimportant at tip speeds below a Mach number of 0.2 which can be expected of a ROH compressor.

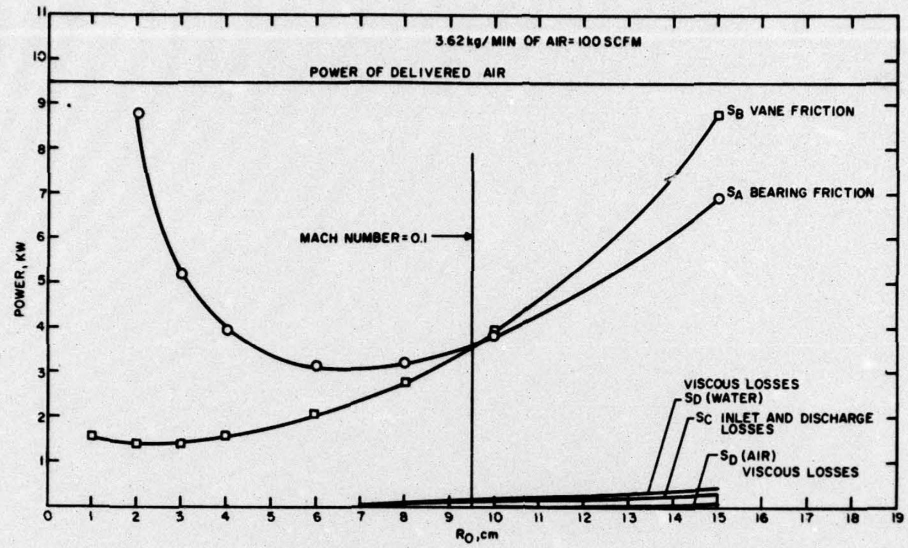

Figure 5-B
Mechanical Losses of Vane Compressor
for 3,600 R/m, 3.62 Kg/Min Operation

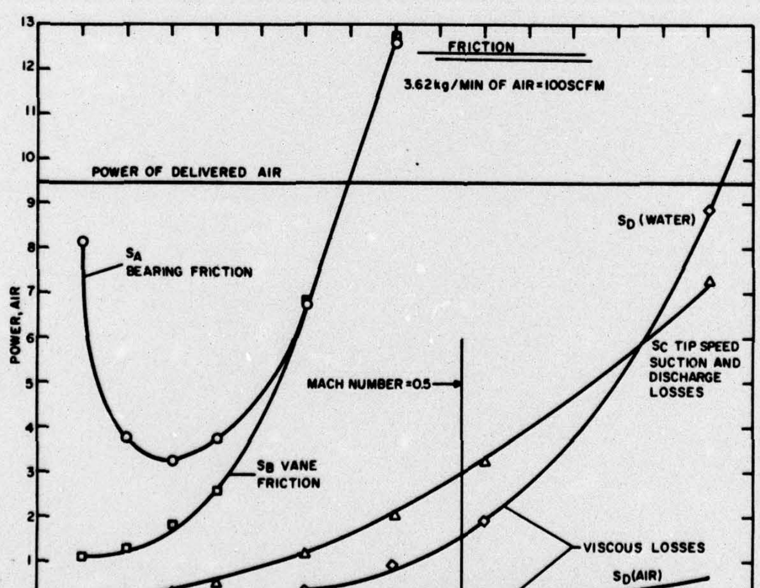

Figure 6-B
Mechanical Losses of Vane Compressor
for 18,000 R/m, 3.62 Kg/Min Operation

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